

Thermodynamic evaluation of a second order simulation for Yoke Ross Stirling engine

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ABSTRACT

Environmental impact and depletion of mineral resources such as coal, oil and gas are prompting a reexamination of an alternative to these resources. A safe and sustainable energy pathway which is crucial to sustainable development in addition to greenhouse gas emitters and its relationship with climate change are leading factors to look for adequate strategies concerning both energy saving and environmental protection. Solar heat engines are attracting much interest nowadays and, as a consequence, different Stirling engine coupled to solar collector have been investigated since it meets the demands of the efficient use of energy and assuring environmental security. In recent years several prototypes and experimental facilities of solar Stirling engine have been developed. The future commercial interest of this alternative for electric power generation relies on a reduction of investment costs and on an increase of performance. The Stirling heat engine was first patented in 1816 by Robert Stirling. Since then, several Stirling engines based on his invention have been built in many forms and sizes. The engine works with a closed cycle and uses several gases as working fluid. Several prototypes have already been studied and produced but the alpha Stirling engine using the Ross Yoke linkage was not well studied although this kind of engine has the advantage of minimizing lateral forces acting on the pistons and leading to a more efficient and compact design compared to beta or gamma Stirling configuration. The objective of this work was the study of the effect of the geometrical and physical parameters on Ross Yoke Stirling engine performance in order to determine the significant thermodynamic parameters having an impact on engine performance. We proposed thermodynamic optimization of a Ross Yoke Stirling engine on the basis of a numerical model integrating the internal and external irreversibility. As a result, this analysis indicated that the performance of a Ross Yoke Stirling cycle engine with air as working gas depends critically on the geometrical parameters and heat input.

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1. Introduction

Reduction of Greenhouse Gas (GHG) emissions can mainly be achieved through utilizing the abundant Renewable Energy Sources (RESs) and the implementation of Energy Efficiency (ENEf) measures. Renewable energy, with the availability of its renewability and non-pollution, will grow to be an effective and practical choice to guarantee the future development of the world.

Developing renewable energy is an inevitable choice for sustainable economic growth, for the harmonious coexistence of human and environment as well as for the sustainable development. Solar energy is one of the more attractive renewable energy sources, especially in MENA countries, and can be used as an input energy source for Stirling engine representing one of the most effective applications [1]. Stirling engine is a regenerative

externally heated engine operating with a cycle that has the same thermal efficiency with Carnot cycle. Since it can be powered by various heat sources and is also able to use solar radiation as energy source. Theoretical and experimental numerous studies have been conducted on Stirling engines. Many types have been designed and built [2–10] however alpha Stirling engine using the Ross Yoke linkage was not well presented and studied although its advantage of minimizing lateral forces acting on the pistons and leading to a more efficient and compact design compared to others configuration.

None of Stirling engine became competitive with the internal combustion engines due to the irreversibility effects caused by external heating and imperfect regeneration. Several studies and prototypes were produced, in solar energy investigations. The first use of Stirling engine using solar energy was made by Parker and Malik in 1962. In their conversion system, solar radiation was concentrated and reflected on to the heater of the engine by using a Fresnel lens.

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Nomenclature

A	area (m^2)
C_p	specific heat at constant pressure ($\text{J kg}^{-1} \text{K}^{-1}$)
C_{pr}	heat capacity of each cell matrix (W K^{-1})
C_v	specific heat at constant volume ($\text{J kg}^{-1} \text{K}^{-1}$)
d	hydraulic diameter (m)
D	diameter (m)
ε	regenerator efficiency
f_r	friction factor
J	annular gap between displacer and cylinder (m)
G	working gas mass flux ($\text{kg m}^{-2} \text{s}^{-1}$)
k	thermal conductivity ($\text{W m}^{-1} \text{K}^{-1}$)
L	length (m)
M	mass of working gas in the engine (kg)
$\dot{m}$	mass flow rate (kg s^{-1})
m	mass of gas in different component (kg)
P	pressure (Pa)
Q	heat (J)
$\dot{Q}$	power (W)
R	gas constant ($\text{J kg}^{-1} \text{K}^{-1}$)
T	temperature (K)
U	convection heat transfer coefficient ($\text{W m}^{-2} \text{K}^{-1}$)
V	volume (m^3)
W	work (J)
Z	displacer stroke (m)

Subscripts

c	compression space
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ch	load
cd	conduction
d	expansion space
E	entered
ext	outside
f	cooler
h	heater
irr	irreversible
moy	average
p	loss
Pa	wall
pis	piston
r	regenerator
$r1$	regenerator cell 1
$r2$	regenerator cell 2
S	left
$shtl$	shuttle

Greek letters

θ	crank angle (rad)
μ	working gas dynamic viscosity ($\text{kg m}^{-1} \text{s}^{-1}$)
ρ	density (kg m^{-3})
ω	angular frequency (rad s^{-1})
γ	$C_p C_v^{-1}$

Popescu [11] studied the influence of the losses of heat and irreversibility on the performance of the engine by finite-time thermodynamic optimization of an endo- exo-irreversible Stirling motor. He demonstrated that the most important reduction of the performances is due to the non-adiabatic regenerator.

Kolin [12] studied several mechanisms of the piston movement transformation and showed that more the movement is discontinuous more the real cycle approaches to the ideal cycle.

In order to obtain a compact Stirling engine and with reduced price, Hirata [13] developed a prototype using the air as working fluid. He measured the mechanical losses for various values of load pressure to obtain the highest output. Karabulut and Koca [14] developed a Stirling engine with swept volume equal to 260 cm^3 . They studied the effect of load pressure and the heating temperature on the torque. The maximum capacity of engine equal to 65 W was obtained for the heating temperature 1100°C and with the load pressure of 2, 5 bars.

Formosa and Despesse [15] developed an analytical model for a free piston Stirling engine in order to investigate the effects of the technological and operating parameters on Stirling engine performance. He deduced that the cooler effectiveness affects the engine performances.

Chin-Hsiang et al. [16] developed a numerical model for a beta-type Stirling engine with rhombic-drive mechanism by taking into account different thermal losses. They predicted all thermodynamic parameters. They concluded that the performance could be improved by adjusting the influential parameters including the regenerative gap, the distance between the two gears, the offset distance from the crank to the center of the gear, and the heat source temperature.

Karabulut [17] developed a numerical model to study a free piston Stirling engine working with closed and open thermodynamic cycle. He concluded that the power exhibits very big variations with respect to the static position of the piston and the displacer,

the hot end temperature, the piston damping coefficient, the displacer rod diameter and the stiffness of the piston spring.

Cinar et al. [18] carried out an experimental study on a beta-type Stirling engine working at atmospheric pressure. He concluded that an increase in the hot-source temperature leads to an increase in the engine speed and torque and the power-output. Besides, the performance can be increased by using a working fluid with a higher thermal conductivity, such as helium or hydrogen.

Cullen and McGovern [19] developed a model for theoretical decoupled Stirling cycle engine used for the analysis of the ideal Stirling cycle engine and its limits on its real-world realization.

Thombare and Verma [20] provided a literature review of the important efforts carried out to develop the Stirling engine cycle and techniques used for engine analysis. They concluded that in order to achieve successful operation of the engine system with good efficiency, a careful design of heat exchangers with a proper selection of the drive mechanism and the engine configuration are crucial. Their study indicated that a Stirling cycle engine working with a relatively low temperature with air as working fluid is potentially the attractive engines of the future, especially solar-powered low-temperature differential Stirling engines with vertical, double acting and gamma configuration.

Karabulut et al. [21] conducted an experimental study of a Stirling engine with a lever-controlled displacer driving mechanism charged with helium. He concluded that the effect of charge pressure on the performance characteristics is positive up to a certain value and then negative. The relation between the hot end temperature and the power seems to be linear. For low and moderate heat sources including solar energy, the engine promises a reasonable performance.

Kongtragool and Wongwises [22] designed and constructed a two single-acting, twin power piston and four power pistons with gamma-configuration and low temperature differential Stirling engine. They tested the engine with various heat inputs. The

variation of the engine torque, the shaft power and the brake thermal efficiency at various heat inputs with engine speed and performance were presented. They concluded that the engine performance increases while the heat input rises. The engine torque, the shaft power, the brake thermal efficiency, the speed, and the heater temperature also increase with the increasing heat input. Besides, the Beale number of this engine increases in parallel with the decreasing temperature ratio or with the increasing heater temperature.

Puech and Tishkova [23] conducted a theoretical investigation concerning the thermodynamic analysis of Stirling engine with linear and sinusoidal variations of the volume. They demonstrated that the engine efficiency with perfect regeneration did not depend on the regenerator dead volume. In fact, this latter amplified the imperfect regeneration effect.

Wongwises and Kongtragool [24] carried out a study on the power output determination of a gamma-configuration, low temperature differential Stirling engine. They demonstrated that the mean pressure power formula is most appropriate for the calculation of a gamma-configuration, low temperature differential Stirling engine power output. Ross Yoke Stirling engine has many known advantages such as minimizing vibration and lateral forces acting on the pistons. Which lead to a more efficient and compact design with insignificant mechanical and thermal losses compared to other Stirling engines. In the present work, an attempt has been made to study this type of engine in order to improve its performance.

According to the performed literature survey [25,15,26–30], Stirling engine of β -type produces the highest indicated work whereas the γ -type produces the lowest indicated work. Whereas, alpha Stirling engines drawn the minimum attention due to an old believe that it has low performance. So, there are few literatures concerning with the performance of this type of engines. However, it has many known advantages such as minimizing vibration and lateral forces acting on the pistons. Which lead to a more efficient and compact design with insignificant mechanical and thermal losses compared to other Stirling engines. In the present work, an attempt has been made to predict the performance behaviors of Ross-yoke Stirling engine in order to improve its performance.

A detailed study of the influence of the Stirling engine design variables seems to be very useful. It will allow to optimize the different engine components and to increase their performance. That's why we developed a calculating code which is based on the dynamic model taking into account the losses in the different elements of the engine. This study demonstrated that a good optimization of the regenerator is necessary for this type of Stirling engine.

2. Stirling engine classification

2.1. Operating principle

The engine integrated two volumes at variable temperatures related together by a regenerative heat exchanger and auxiliary heat exchangers in order to realize the thermodynamic Stirling cycle.

According to their geometric configuration, Stirling engines were basically classified into types. In fact, there are three configuration systems: Alpha, Beta, and Gamma as shown in Fig. 1.

These conditions described only the Stirling engine cylinder couplings which recognized the mode in which the displacer piston and the power piston were linked, with respect to the connection of the variable working spaces volume. These represent the spaces inside the engine cylinder where the working fluid is heated and cooled respectively in a closed volume.

2.2. The different configurations of Stirling engines

The majority of the literature concerning the Stirling engine was published sporadically since 1816 and there were no standardization or clarification of this terminology. This led to rely on diesel engine more than Stirling engine. A rich variety of literature engines is available, but no standard terminology concerning the best configuration exists.

The Beta engine [31] includes the set of Stirling engine with a single cylinder pact and where the displacer and the power piston are linked in tandem. The power is generated by the action of the pistons jointly. This type of configuration has sealing problems. Besides, it loses its advantages compared to multi-cylinder engines.

The Gamma engine [32] is very comparable to beta-group since the power output is generated in the same way as in Beta engines. The only disparity is the fact that the two pistons move in separate cylinders. The considerable dead space that this configuration has represents the disadvantage of this group.

The alpha engine [29] consists of two separate cylinders that each one has its own piston, either the displacer or the power piston and three heat exchangers. Each heat source is linked to its own cylinder. The power output is produced by the separate motion of the individual piston.

The advantages of Alpha Stirling engine using the Ross yoke linkage (Fig. 2) are well known [33] because of the high power-to-volume ratio. However, the thermodynamic analysis of this engine was so restricted according to a literature overview since it is reported to necessitate high temperature but this problem was solved after the latest technological advances.

Drive mechanisms cause a complicated problem for Stirling engines as discontinuous motion is necessary to achieve the volumetric changes that lead to a net power output [34].

3. Losses in a Stirling engine

The energy loss in a Stirling engine is due to the thermodynamic processes and to the mechanical losses. The Compression and the expansion are not adiabatic. Load losses exist in the heat exchangers which are not ideal.

3.1. Power loss by pressure drop $\delta\dot{Q}_{Pch}$

Pressure drops due to friction and to area changes in the heat exchangers can easily be modeled using steady flow correlations. These pressure drops lead to a dissipation of work within the heat exchangers leading to a reduction of the available work in the machine.

The frictional drag force is expressed by [35]:

$$F = \frac{2f_f G^2 V}{d\rho} \quad (1)$$

where G is working gas mass flux ($\text{kg m}^{-2} \text{s}^{-1}$), d is the hydraulic diameter, ρ is gas density (kg m^{-3}), V is volume (m^3) and f_f is the conventional fanning friction factor.

A more suitable definition of the friction factor was proposed by Ureili [36], being so-called 'Reynolds friction factor', as follows:

$$f_r = f_f Re \quad (2)$$

Re is the Reynolds number

$$Re = \frac{Gd}{\mu} \quad (3)$$

The frictional drag force must be equal and opposite to the pressure drop force:

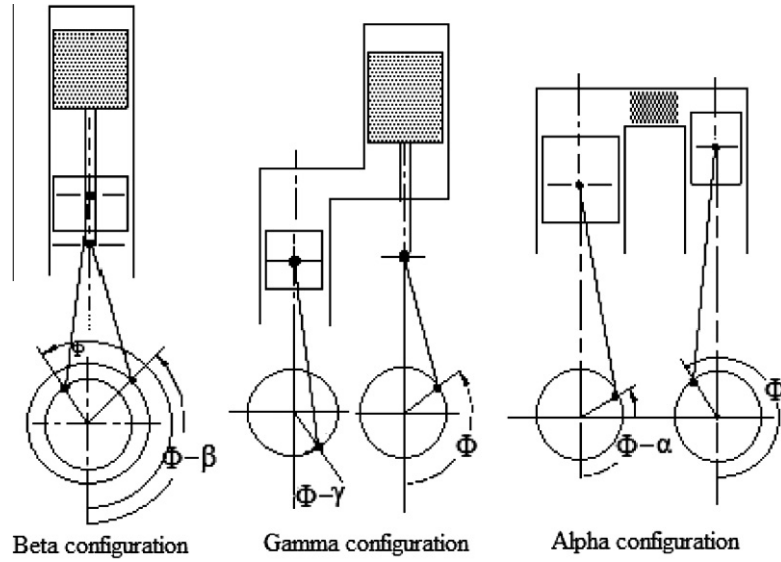


Fig. 1. Stirling engine configuration.

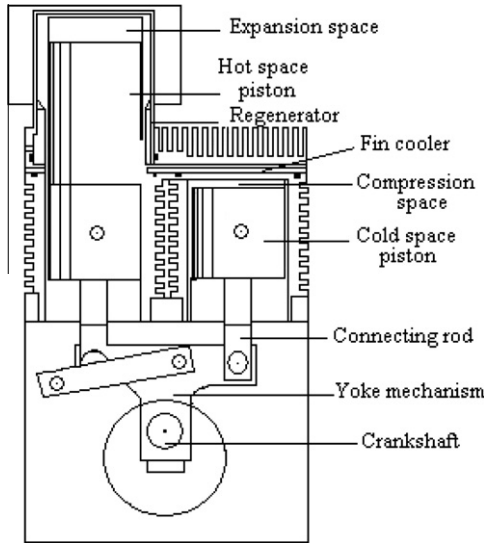


Fig. 2. The Ross Yoke drive engine-schematic cross section view.

$$F + \Delta pA = 0 \quad (4)$$

Substituting Eqs. (2) and (3) into Eq. (1) we finally obtain:

$$\Delta p = -\frac{2f_r \mu G V}{A d^2 \rho} \quad (5)$$

The internal heat generation which occurs when the gas is forced to flow against the frictional drag force, is given by [37]:

$$\delta \dot{Q}_{Pch} = -\frac{\Delta p \dot{m}}{\rho} \quad (6)$$

$\dot{m}$ is the mass flow rate (kg s^{-1}).

The total heat generated by pressure drop in the different exchangers is:

$$\delta \dot{Q}_{PchT} = \delta \dot{Q}_{Pchf} + \delta \dot{Q}_{Pchr1} + \delta \dot{Q}_{Pchr2} + \delta \dot{Q}_{Pchh} \quad (7)$$

where $\delta \dot{Q}_{Pchf}$ is the heat generated by pressure drop in the cooler, $\delta \dot{Q}_{Pchr1}$ is the heat generated by pressure drop in the regenerator

$r1$, $\delta \dot{Q}_{Pchr2}$ is the heat generated by pressure drop in the regenerator $r2$ and $\delta \dot{Q}_{Pchh}$ is the heat generated by pressure drop in the heater.

3.2. Energy Loss by internal conduction $\delta \dot{Q}_{Pcd}$

Energy losses due to the internal thermal conduction between the hot parts and the cold parts of the engine through the solid matrix of the exchangers are taken into account. These losses are directly proportional to the temperature difference between the two ends of the exchanger; they are given for the different exchangers [38]:

$$\delta \dot{Q}_{Pcdr} = k_{cdr} \frac{A_r}{L_r} (T_{r-h} - T_{f-r}) \quad (8)$$

$$\delta \dot{Q}_{Pcdf} = k_{cdf} \frac{A_f}{L_f} (T_{f-r} - T_{c-f}) \quad (9)$$

$$\delta \dot{Q}_{Pcdh} = k_{cdh} \frac{A_h}{L_h} (T_{h-d} - T_{r-h}) \quad (10)$$

With $\delta \dot{Q}_{Pcdr}$, $\delta \dot{Q}_{Pcdf}$ and $\delta \dot{Q}_{Pcdh}$ representing respectively the conduction loss in the regenerator, the cooler and the heater.

The total conduction loss is then:

$$\delta \dot{Q}_{PcdT} = \delta \dot{Q}_{Pcdr} + \delta \dot{Q}_{Pcdf} + \delta \dot{Q}_{Pcdh} \quad (11)$$

To reduce the amount of energy lost from the system by internal conduction, the k_{cdr} , A_r , L_r and ΔT parameters should be optimized.

3.3. Loss by external conduction

The loss by external conduction is considered in the regenerator which is not adiabatic compared to the external side whereas for the other exchangers, the external losses are negligible. The energy stored by the regenerator at the time of the passage of gas from the space of expansion to the space of compression is not completely restored to this gas at the time of its return. So energy losses towards outside exist; they are determined from the regenerator effectiveness given by the following equation [39]:

$$E = \frac{NUT}{1 + NUT}$$

The energy lost by external conduction in the regenerator is so:

$$\delta \dot{Q}_{\text{Pext}} = (1 - E) \delta \dot{Q}_r \quad (12)$$

With $\delta \dot{Q}_r = \delta \dot{Q}_{r1} + \delta \dot{Q}_{r2}$

Q_r : exchanged energy in the regenerator.

The reduction of the energy lost by external conduction is very important since it represents the most significant loss in the Stirling engine. It depends on the regenerator effectiveness.

3.4. Loss by shuttle effect $\delta \dot{Q}_{\text{Pshtl}}$

The losses by shuttle effect are due to the piston reciprocal action in the cylinder. The piston absorbs a heat quantity from the hot source and restores it to the cold source. Martini [37] proved that:

$$\delta \dot{Q}_{\text{Pshtl}} = \frac{0.4Z^2 k_{\text{pis}} D_d}{J L_d} (T_d - T_c) \quad (13)$$

4. Design specification and concept

4.1. Engine specification

In order to avoid losses and to obtain high thermal efficiency the engine parameters should be optimized. The working fluid is air and the temperature difference between the heater and the cooler is about 600 °C only.

The engine presented in Fig. 2 uses a Ross Yoke mechanism driving two pistons by the mean of yoke linkage [37]. The major feature of this is that the Ross Yoke is an ingenious mechanism for transferring dual piston motion into rotational motion. It has the advantage of minimizing lateral forces acting on the pistons leading to a more efficient and compact design.

4.2. Design concept

The yoke drive mechanism does not generate sinusoidal volume variations. It is reported that the exact piston displacement functions are very intricate.

The volume variations are derived from geometric considerations in Fig. 3 and Table 1. The major design concepts are listed in Table 2.

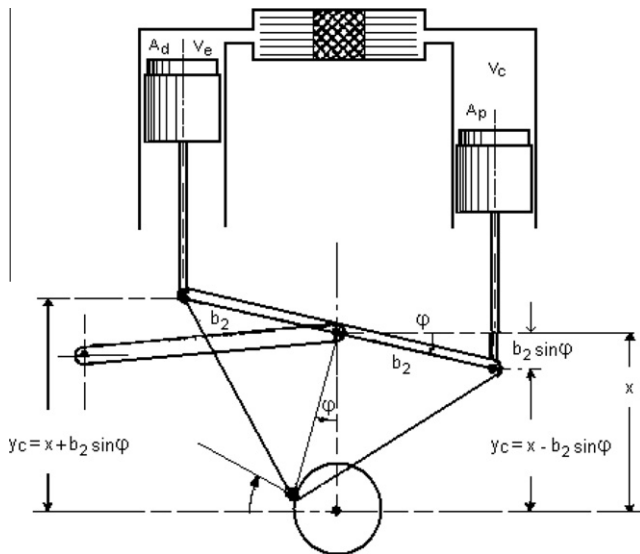


Fig. 3. Geometric derivation of the Ross Yoke drive equation.

Table 1
Volumes variations.

Geometrical parameters	$b_1 \sin \phi = r \cos \theta$ $b \theta = \sqrt{b_1^2 (r - \cos \theta)^2}$ $X = r \sin \theta + b_e$
Displacements	$Y_c = r[\sin \theta - \cos \theta (b_2/b_1)] + b_\theta$ $Y_e = r[\sin \theta + \cos \theta (b_2/b_1)] + b_e$
Volume variations	$V_c = V_{mc} + A_p(Y_{\max} - Y_c)$ $V_e = V_{me} + A_d(Y_{\max} - Y_e)$
$\frac{dV_c}{d\theta} = -A_p r \left[\cos \theta + \sin \theta \left(\frac{b_2}{b_1} \right) + \frac{r \sin \theta \cos \theta}{b_\theta} \right]$	
$\frac{dV_e}{d\theta} = -A_d r \left[\cos \theta - \sin \theta \left(\frac{b_2}{b_1} \right) + \frac{r \sin \theta \cos \theta}{b_e} \right]$	

Table 2
Concepts and target performance.

Parameters	Values/type
Engine type	Alpha
Working fluid	Air
Crank length	$r = 7.6$ mm
Yoke crank length	$b_1 = 29$ mm
Piston length	$b_2 = 29$ mm
Displacement extremities	$Y_{\min} = 17.75$ mm $Y_{\max} = 39.28$ mm
Swept volumes	$V_{bc} = V_{mc} = 24.42$ cm ³
Heat exchangers volumes	$V_c = 5$ cm ³ $V_r = 6.75$ cm ³ $V_h = 4.78$ cm ³
Mean phase angle advance	$\alpha = 90^\circ$
Mass of gas in engine	$M = 1$ g
Hot space temperature	$T_h = 923$ K
Cold space temperature	$T_c = 350$ K
Frequency	Freq = 41.72 Hz

5. Dynamic model of Stirling engine

There are many different ways to degrade the power produced by an ideal machine and in order to accurately predict the power and the efficiency; a clear understanding of the design compartments is required.

A mathematical model takes into consideration the pressure drop in the heat exchangers [40–47].

Heat transfer and flow friction in the heat exchangers, i.e. the heater, the cooler and the regenerator, are evaluated using empirical equations under steady flow condition.

No leakage is allowed either through the appendix gap or through the seals of the connecting rods.

The gas temperature in the various engine compartments is variable.

The cooler and the heater walls are maintained isothermally at temperatures T_{paf} and T_{pah} .

5.1. Dynamic model Equations

The gas temperatures in the various compartments are calculated from the perfect gas law:

$$T_c = \frac{P_c \cdot V_c}{R \cdot m_c} \quad (14)$$

$$T_f = \frac{P_f \cdot V_f}{R \cdot m_f} \quad (15)$$

$$T_h = \frac{P_h \cdot V_h}{R \cdot m_h} \quad (16)$$

$$T_d = \frac{P_d \cdot V_d}{R \cdot m_d} \quad (17)$$

The regenerator is divided into two cells $r1$ and $r2$, each cell being associated with its respective mixed mean gas temperature T_{r1} and T_{r2} expressed as follow:

$$T_{r1} = \frac{P_{r1} \cdot V_{r1}}{R \cdot m_{r1}} \quad (18)$$

$$T_{r2} = \frac{P_{r2} \cdot V_{r2}}{R \cdot m_{r2}} \quad (19)$$

An extrapolated linear curve is drawn through temperature values T_{r1} and T_{r2} defining the regenerator interface temperature T_{r-f} , T_{r-r} and T_{r-h} , as follows [32]:

$$T_{r-f} = \frac{3T_{r1} - T_{r2}}{2} \quad (20)$$

$$T_{r-r} = \frac{T_{r1} + T_{r2}}{2} \quad (21)$$

$$T_{r-h} = \frac{3T_{r2} - T_{r1}}{2} \quad (22)$$

According to the flow direction of the fluid, the interface's temperatures: T_{c-f} , T_{f-r} , T_{r-h} and T_{h-d} are defined as follows [32]:

If $\dot{m}_{c-f} > 0$ then $T_{c-f} = T_c$ otherwise $T_{c-f} = T_f$

If $\dot{m}_{f-r} > 0$ then $T_{f-r} = T_f$ otherwise $T_{f-r} = T_{r-f}$

If $\dot{m}_{r-h} > 0$ then $T_{r-h} = T_{r-h}$ otherwise $T_{r-h} = T_h$

If $\dot{m}_{h-d} > 0$ then $T_{h-d} = T_h$ otherwise $T_{h-d} = T_d$

where T_{c-f} is the temperature of the interface between compression space and the cooler, T_{f-r} is the temperature of the interface between the cooler and regenerator, T_{r-h} is the temperature of the interface between the regenerator and the heater, T_{h-d} is the temperature of the interface between the heater and the expansion space.

The matrix temperatures are thus given by:

$$\frac{dT_{pa_{r1}}}{dt} = -\frac{\delta Q_{r1}}{c_{pr} dt} \quad (23)$$

$$\frac{dT_{pa_{r2}}}{dt} = -\frac{\delta Q_{r2}}{c_{pr} dt} \quad (24)$$

where C_{pr} is the heat capacity of each cell matrix ($J K^{-1}$), Q_{r1} is the quantity of heat exchanged to the regenerator $r1$ (J), Q_{r2} is the quantity of heat exchanged to the regenerator $r2$ (J), $T_{pa_{r1}}$ is the matrix temperatures in the regenerator $r1$ (K) and $T_{pa_{r2}}$ is the matrix temperatures in the regenerator $r2$ (K).

By taking into account the conduction loss in the exchangers and the regenerator effectiveness, the power exchanged in the various exchangers is written:

$$\delta \dot{Q}_f = h_f A_{paf} (T_{paf} - T_f) \quad (25)$$

$$\delta \dot{Q}_{r2} = h_{r2} A_{par2} (T_{par2} - T_{r2}) \quad (26)$$

$$\delta \dot{Q}_{r1} = h_{r1} A_{par1} (T_{par1} - T_{r1}) \quad (27)$$

$$\delta \dot{Q}_h = h_h A_{pah} (T_{pah} - T_h) \quad (28)$$

The heat transfer coefficient of exchanges h_f , h_{r1} , h_{r2} and h_h are only available empirically [29].

The total exchanged heat is:

$$\delta \dot{Q} = \delta \dot{Q}_f + \delta \dot{Q}_{r1} + \delta \dot{Q}_{r2} + \delta \dot{Q}_h \quad (29)$$

The work given by the cycle is:

$$\frac{\delta W}{dt} = P_c \frac{dV_c}{dt} + P_d \frac{dV_d}{dt} \quad (30)$$

The thermal efficiency given by the cycle is:

$$\eta = \frac{W}{Q_h} \quad (31)$$

The total engine volume is:

$$V_T = V_c + V_f + V_{r1} + V_{r2} + V_h + V_d$$

The other variables of the dynamic model are given from the energy and mass conservation equation, applied to a generalized cell as follows:

Energy conservation equation [33,34]:

$$\delta \dot{Q} + C_p T_E \dot{m}_E - C_p T_S \dot{m}_S = P \frac{dV}{dt} + C_v \frac{d(mT)}{dt} \quad (32)$$

Since there is a varying pressure distribution throughout the engine, we have arbitrarily chosen the compression space pressure P_c as the baseline pressure. Thus at each increment of the solution, P_c will be evaluated from the relevant differential equation and the pressure distribution determined with respect to P_c . Thus it can be obtained from the following expression:

$$P_f = P_c + \frac{\Delta P_f}{2} \quad (33)$$

$$P_{r1} = P_f + \frac{(\Delta P_f + \Delta P_{r1})}{2} \quad (34)$$

$$P_{r2} = P_{r1} + \frac{(\Delta P_{r1} + \Delta P_{r2})}{2} \quad (35)$$

$$P_h = P_{r2} + \frac{(\Delta P_{r1} + \Delta P_h)}{2} \quad (36)$$

$$P_d = P_h + \frac{\Delta P_h}{2} \quad (37)$$

Applying energy conservation equation to the various engine cells, we obtain:

$$-C_p T_{c-f} \dot{m}_c = \frac{1}{R} \left(C_p P_c \frac{dV_c}{dt} + C_v V_c \frac{dP_c}{dt} \right) \quad (38)$$

$$\delta \dot{Q}_f - \delta \dot{Q}_{pchl} + C_p T_{c-f} \dot{m}_{fe} - C_p T_{f-r} \dot{m}_{fs} = \frac{C_v V_f}{R} \frac{dP_c}{dt} \quad (39)$$

$$\delta \dot{Q}_{r1} - \delta \dot{Q}_{pchr1} + C_p T_{f-r} \dot{m}_{r1E} - C_p T_{r-r} \dot{m}_{r1S} = \frac{C_v V_{r1}}{R} \frac{dP_c}{dt} \quad (40)$$

$$\delta \dot{Q}_{r2} - \delta \dot{Q}_{pchr2} + C_p T_{r-r} \dot{m}_{r2E} - C_p T_{r-h} \dot{m}_{r2S} = \frac{C_v V_{r2}}{R} \frac{dP_c}{dt} \quad (41)$$

$$\delta \dot{Q}_h - \delta \dot{Q}_{pchl} + C_p T_{r-h} \dot{m}_{hE} - C_p T_{h-d} \dot{m}_{hS} = \frac{C_v V_h}{R} \frac{dP_c}{dt} \quad (42)$$

$$C_p T_{h-d} \dot{m}_d = \frac{1}{R} \left(C_p P_d \frac{dV_d}{dt} + C_v V_d \frac{dP_c}{dt} \right) \quad (43)$$

Summing Eqs. (39)–(44) we obtain the pressure variation:

$$\frac{dP_c}{dt} = \frac{1}{C_v \cdot V_T} \left[R(\delta \dot{Q} - \delta \dot{Q}_{pchl}) - C_p \frac{\delta W}{dt} \right] \quad (44)$$

Mass conservation equation:

$$M = m_d + m_c + m_f + m_r + m_h \quad (45)$$

Table 3
Calculations of error analysis.

Cycle iteration	1	2	3	4	5	6	7	8
Power by cycle iteration	23.8807	22.5903	22.2088	22.1731	22.1701	22.1698	22.1698	22.1698
Efficiency by cycle iteration	0.3760	0.3030	0.2949	0.2942	0.2942	0.2942	0.2942	0.2942
Error for power, ε_p	1.2904	0.3815	0.0387	0.003	0.003	0.0000	0.0000	0.0000
Error for efficiency, ε_e	0.0730	0.3428	0.0081	0.0000	0.0000	0.0000	0.0000	0.0000

The mass flow in the various engine compartments are given from the conservation equations of energy (39)–(44):

$$\dot{m}_{cS} = -\frac{1}{RT_{c-f}} \left(P \frac{dV_c}{dt} + V_c \frac{dP_c}{\gamma dt} \right) \quad (46)$$

$$\dot{m}_{fS} = \frac{1}{c_p T_{f-r}} (\delta \dot{Q}_f - \delta \dot{Q}_{pchf} + c_p T_{c-f} \dot{m}_{fE} - \frac{c_v V_f}{R} \frac{dP_c}{dt}) \quad (47)$$

$$\dot{m}_{r1S} = \frac{1}{c_p T_{r-r}} (\delta \dot{Q}_{r1} - \delta \dot{Q}_{pchr1} + c_p T_{f-r} \dot{m}_{r1E} - \frac{c_v V_{r1}}{R} \frac{dP_c}{dt}) \quad (48)$$

$$\dot{m}_{r2S} = \frac{1}{c_p T_{r-h}} (\delta \dot{Q}_{r2} - \delta \dot{Q}_{pchr2} + c_p T_{r-r} \dot{m}_{r2E} - \frac{c_v V_{r2}}{R} \frac{dP_c}{dt}) \quad (49)$$

$$\dot{m}_{hS} = \frac{1}{c_p T_{h-d}} (\delta \dot{Q}_h - \delta \dot{Q}_{pchh} + c_p T_{r-h} \dot{m}_{hE} - \frac{c_v V_h}{R} \frac{dP_c}{dt}) \quad (50)$$

where $\dot{m}_{cS} = \dot{m}_{fE}$; $\dot{m}_{fS} = \dot{m}_{r1E}$; $\dot{m}_{r1S} = \dot{m}_{r2E}$; $\dot{m}_{r2S} = \dot{m}_{hE}$ and $\dot{m}_{hS} = \dot{m}_{dE}$

5.2. Method of solution

The Stirling engine tested has a Ross Yoke linkage system, as shown in Fig. 2. The geometrical parameters of this engine are given in Table 1.

The operating conditions are as follows: working gas: air at a mean pressure of 2.195 bar; frequency: 41.72 Hz; hot space temperature: $T_{pah} = 923$ K; cold space temperature: $T_{pad} = 350$ K.

The dynamic model developed in this work including thermal irreversibilities was applied and tested on the Ross Yoke Stirling engine.

The independent differential equations obtained in paragraph 4, are solved simultaneously for the variables P_c , m_c , T_{r1} , W , etc.

The problem thus reduces to the simultaneous solution of a set of ordinary differential equations. The simplest approach to solve this set of ordinary differential equations is to formulate it as an initial-value problem, where the initial values of all the variables are known and the equations are integrated from this initial state over a complete cycle, bringing the pistons back to their initial positions.

The corresponding set of differential equations is expressed as $dY/dt = F(t, Y)$. The objective is to find the unknown function $Y(t)$ which satisfies both the differential equations and the initial conditions. By assigning arbitrary initial conditions for the seven variables to be integrated and integrating the equations through several complete cycles, the cyclic steady state can be attained when the respective values at the beginning of the cycle and at the end of the cycle are equal. The residual regenerator heat Q_r at the end of the cycle, should be zero. The system of equations is solved numerically using the classical fourth order Runge–Kutta method, cycle after cycle until periodic conditions are reached. The calculations of error analysis are cited in Table 3.

To validate the numerical method used in the computation, the results are compared with those obtained by Urieli and Berchowitz [36] for the same conditions (adiabatic models) of the GPU-3 engine data. The comparison shows a good agreement, as shown in Fig. 4.

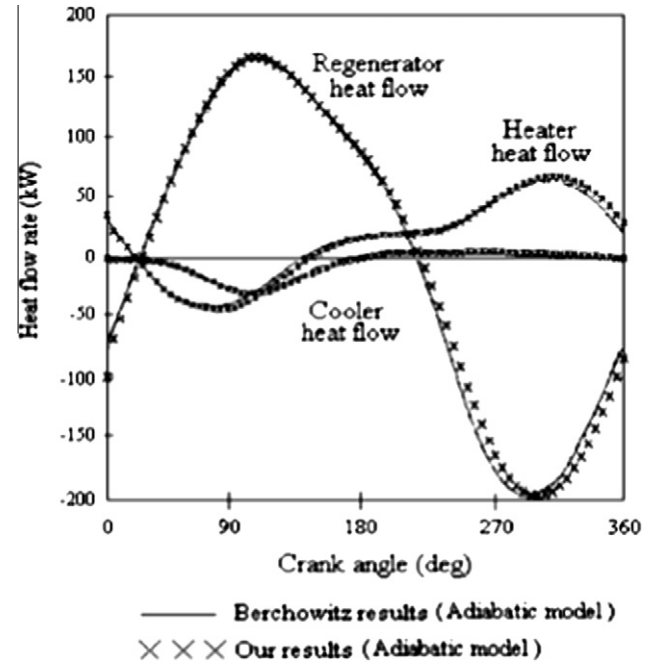


Fig. 4. Validation of the computational method.

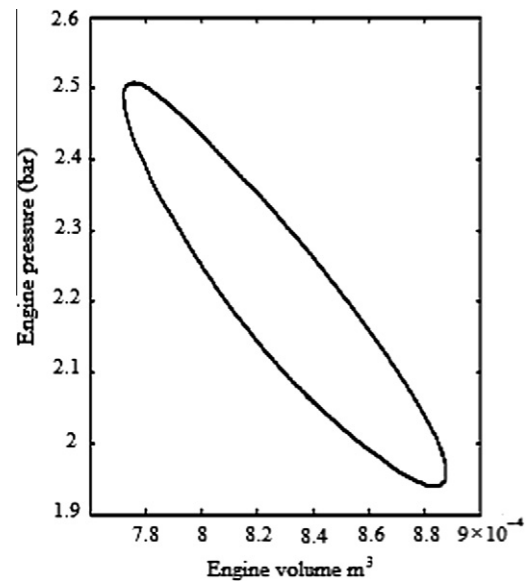


Fig. 5. Pressure volume diagram.

6. Results and discussion

Stirling engines are mechanical devices theoretically based on the Stirling cycle. That is a highly idealized thermodynamic cycle. It consists of four processes which combine to form a closed cycle:

two isothermal and two isochoric processes. The surface of the PV diagram is equal to the network produced by the engine.

The maximal pressure reached in the engine is of 2.507 bars at the end of the compression phase, and the mean pressure is 2.195 bars.

The surface of the PV diagram shown in Fig. 5 is equal to the network produced by the engine.

We notice that the work calculated by the dynamic model is lower than that found by the isothermal model since the PV diagram surface is narrowed. This figure shows also the effect of the sinusoidal movement of the displacer piston on P–V diagram. It is noticed that the average pressure is increased compared to the adiabatic case, due to the effect of temperature variation.

The regenerator is one of the most complicated components of the Stirling engine and is the major limiting factor in Stirling engine performance. It acts as a thermal sponge alternatively accepting heat from working fluid as it passing through it toward the cooler segment and rejecting heat to the working fluid in its way back.

6.1. Heat energy and mass flow in different heat exchangers

Heat exchangers are key components in the Stirling engines which are the heater, regenerator and cooler which exchange heat to and from the engine. The study of the heat transfer behavior within and across the Stirling engine may provide an adequate prediction for overall system thermal efficiency in practice.

Figs. 6 and 7 show the heat energy exchanged and torque variation, according to the dynamic model. Regenerators are typically made up of porous structures, usually referred to as the packing material, which may lead to complex flow and heat transfer behaviors of the working fluid while passing through the regenerator. The heat transferred in the regenerator is very significant almost ten times than that of the net work done per cycle. The heat transfer coefficient is much lower for gases than for liquids, thus the vast surface area in a regenerator greatly increases heat transfer. It is concluded that the engine performance depends critically on the regenerator effectiveness and its ability to accommodate the high heat flow.

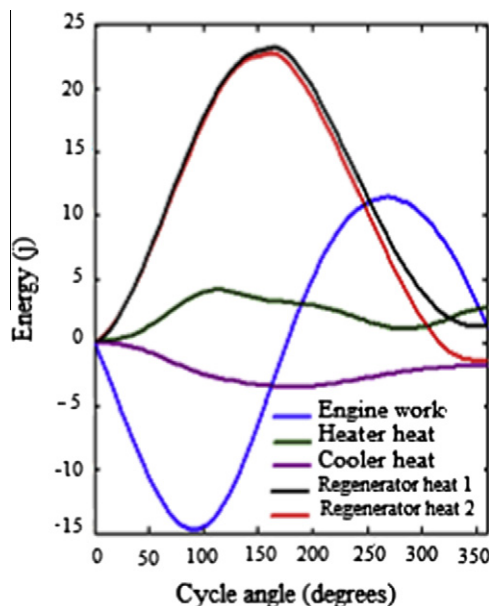


Fig. 6. Heat energy variation.

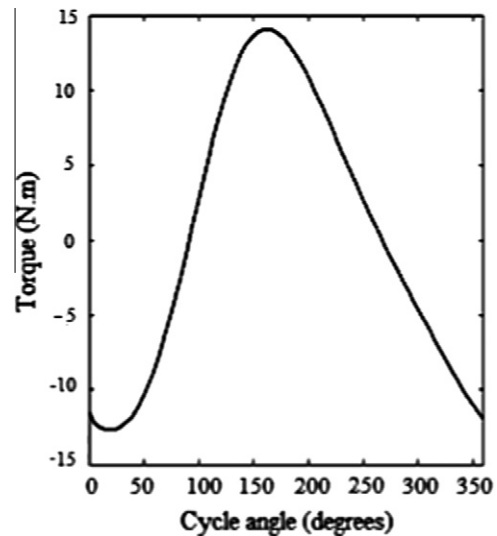


Fig. 7. Torque variation.

When the cycle angle increase the amount of heat exchanged increases until it reaches a maximum equal to 24 J. A regenerator is used to increase the efficiency of a Stirling engine.

Fig. 6 prove also that the regenerator is well designed since the energy rejected by the gas to the regenerator matrix in the first half of the cycle is equal to the energy absorbed by the gas from the matrix in the second half of the cycle. Therefore, High theoretical efficiency associated with Stirling engine compared with internal combustion engine is mainly due to the absence of intake and exhaust process (i.e. no energy exhaust from the Stirling engine).

The power and the thermal efficiency calculated by this model are closer to the real power and thermal efficiency of the prototype than other model. Since the average power is 48.79 W and the engine effectiveness resulted is up to 41%.

It is noted that the flow in the cooler reaches a value greater than the flow in the regenerator which reaches a higher value than the heater. This is due to the location of the cooler that is close to the engine piston.

6.2. Pressure drop and energy dissipation in the different heat exchangers

Stirling engines have pressure losses at heater, regenerator, cooler and their connection parts. The design and configuration of these heat exchangers affect the engine performance. We Notice from Fig. 8 showing the variation of pressure drop in the heater, the regenerator and the cooler that the regenerator is responsible of the undesirable pressure decrease, which attains 0.064 bars because of its geometrical and physical aspects. Whereas the pressure drop in the other exchangers reaches a maximum 0.00268 bar. Large effective heat transfer area and few dead volumes cause high pressure drop across the regenerator. This increase of pressure drop across the regenerator reduces the output power.

Also from Fig. 9 we find that the energy dissipation in the regenerator is so high compared to the other exchangers.

It is also noticed that approximately 51.24% of power is lost by the regenerator compared to the net power of the engine. However its rule is major for the engine output.

6.3. Optimization of the Stirling engine performances

Stirling engines are known as high efficiency engines. In practice due to some reasons their efficiency reduces. Some techniques

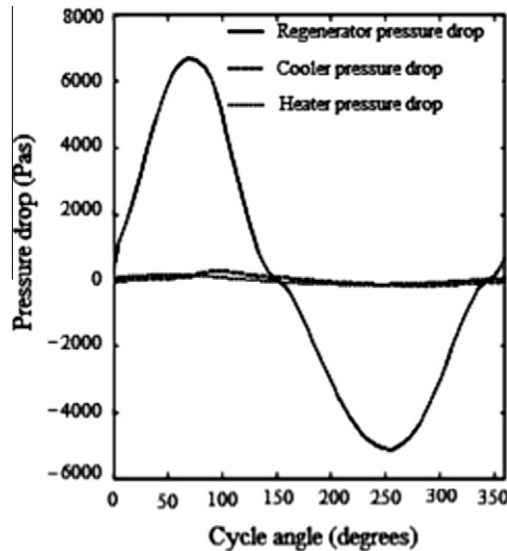


Fig. 8. Pressure drop variation.

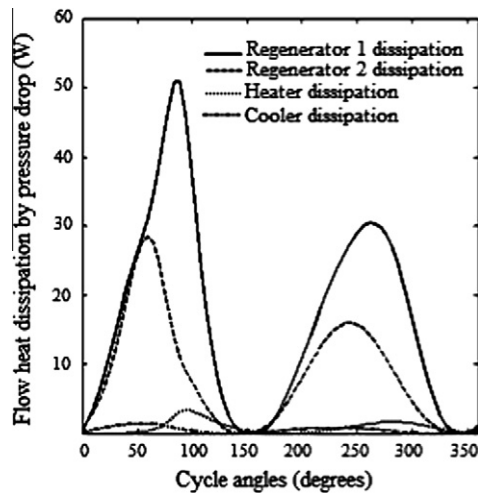


Fig. 9. Dissipation flow in different heat exchangers.

can give an insight into the detailed interaction of factors determining performance, and can be used to produce a significant improvement in the predicted performances of this particular engine. One of the most important sources of efficiency reductions is imperfect heat transfer. The search for an engine cycle with high efficiency has led to improve performance analysis and optimization of Stirling engine which depends on the losses reduction and the determination of the adequate geometrical and physical parameters. In fact, Stirling engine loses efficiency due to large differences in the temperature of the working fluid and the heating and cooling spaces. Usually the design point of a Stirling engine will be somewhere between the two limits of: maximum efficiency point; and maximum power point. Therefore, it is required that for successful operation of engine system with good efficiency a careful design of heat exchangers, proper selection of drive mechanism and engine configuration is essential.

6.3.1. Effect of the heater temperature

The heater transfers heat from external source to the working fluid contained within the engine working space.

Figs. 10 and 11 show the effect of the heater temperature on the engine power and efficiency for different frequencies. It is noted that the engine performances increase according to the heater temperature and the power increases with the operating frequency.

We note that when $T_{pah} < T_{paf}$, the machine can function as a refrigerator; when the expansion temperature decreases, it is then necessary to increase the mechanical power. We demonstrated that we can assemble the machines in two ways: one like a refrigerator and the other like an engine.

The curves indicate that the increase of power with heat source temperature is limitless. Therefore, the progress made in the material technology especially the resistance to high temperature, will improve the performance of Stirling engines.

We notice also that the increase of the heater temperature induces a continuously increase in the heat exchanged.

Although the engine losses increase when the heater temperature rises [13], the performances of the engine augment, this is explained by the increase of the exchanged energy between the matrix and the working fluid of the regenerator. The temperature of working fluid entering the regenerator is not constant because the pressure, density and velocity of working fluid vary over a wide

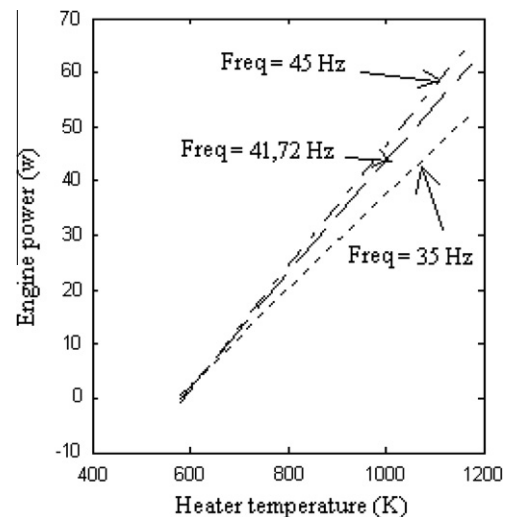


Fig. 10. Temperature effect on engine power.

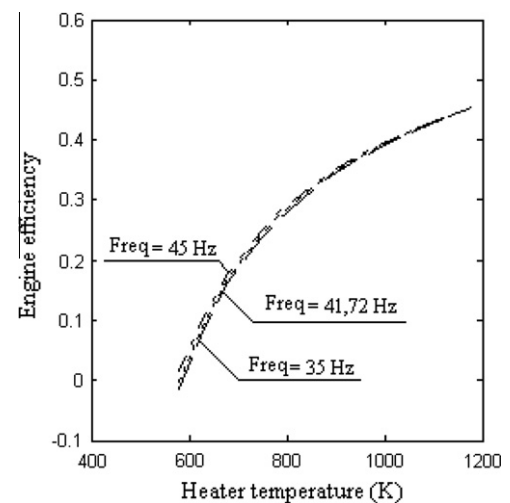


Fig. 11. Temperature effect on engine efficiency.

range. The effectiveness of the regeneration process largely depends upon the thermal capacity of the material also.

6.3.2. Effect of the fluid mass

Figs. 12 and 13 show the fluid mass effect on the engine power and efficiency. It is noted that the power increases according to the mass and that the efficiency reaches a maximum. When the mass increases, the reduction in the output is due to the load loss increase.

An increase of the total mass of gas in the engine leads to a rise in the density, the mass flow, the gas velocity, the load, and pressure function. Therefore, this increase in the total mass of gas in the engine will lead to a more energy loss by pressure drop [16]; however, the engine power increases and the efficiency reaches a maximum value.

The engine efficiency decrease occurring when the mass of fluid increases beyond a certain value is a consequence of the reduction of the maximum temperature of the working fluid secondary to the transfer capacity limitation in the heater and the regenerator. When the mass increases further, the decrease of the efficiency is due to an increase of the pressure loss and the limitation of the heat exchange capacity in the regenerator and the heater.

The two Figs. 12 and 13 show that more we increase the temperature more the engine performance increases and that the optimal value of the fluid mass decreases when temperature increases. It is known that the whole working fluid mass is divided between the cold and hot spaces. The mass also takes part in every process consistently and the process makes up a cycle that includes compression and expansion.

6.3.3. Effect of the swept volume ratio of compression and expansion on the power engine

Figs. 14 and 15 show the effect of the swept volume ratio of compression and expansion ($\tau = V_c/V_d$) on the power and the efficiency. The curves show clearly that there is a well defined optimal value of τ at which the efficiency is maximum. The comparison of the two curves for $(T_{pah}/T_{paf}) = 0.25$ and $(T_{pah}/T_{paf}) = 0.33$ shows that the optimal value is not constant but changes from 0.8 when $(T_{pah}/T_{paf}) = 0.25$ –1.1 when $(T_{pah}/T_{paf}) = 0.33$. The changes of the dead volume ratio produce also an adjustment of the optimal value of τ . Thus, there is not a better fixed value τ . The decrease of the output when τ increases beyond a certain value is probably compared to the instantaneous distribution of the fluid masses in the different exchangers during the cycle. A better mechanical

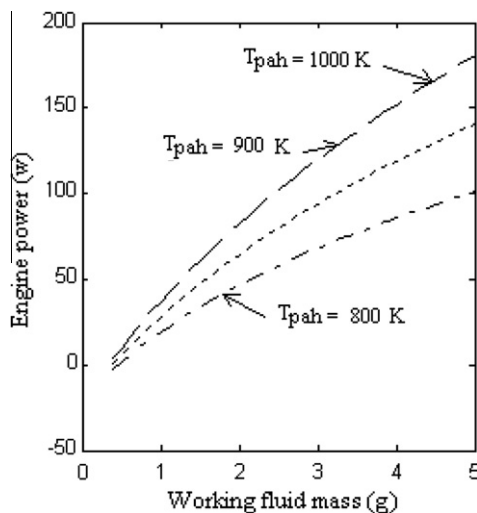


Fig. 12. Effect of working fluid mass on engine power.

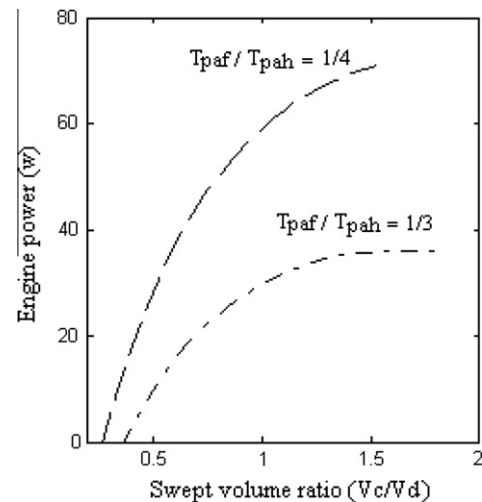


Fig. 14. Effect of the swept volume ratio on the engine power.

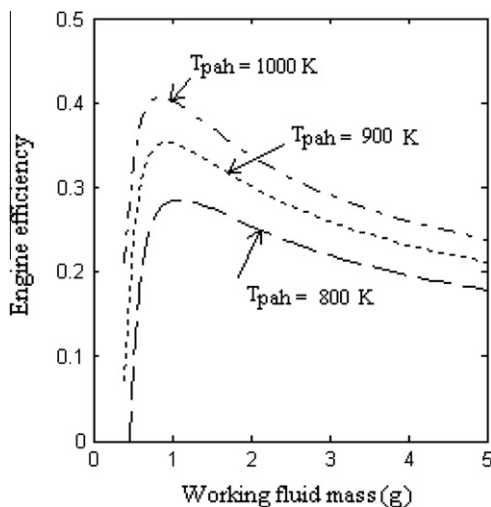


Fig. 13. Effect of working fluid mass on engine efficiency.

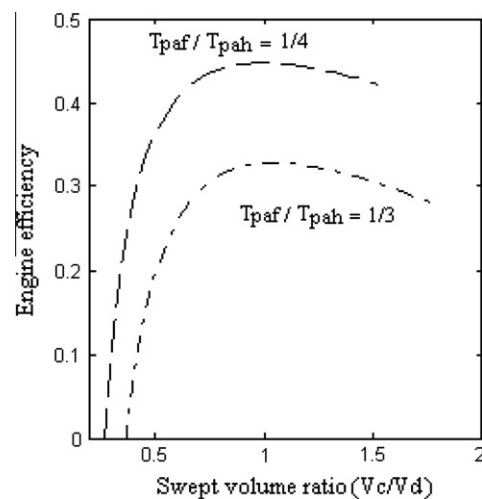


Fig. 15. Effect of the swept volume ratio on the engine efficiency.

efficiency can be obtained by taking a swept volume ratio smaller than an optimal value which vary according to the (T_{pah}/T_{paf}) while resulting in only a relatively small reduction in shaft output.

6.3.4. Effect of the dead volume ratio

Figs. 16 and 17 show the effect of the regenerator volume rate compared to the heater ($\phi = V_r/V_h$) on the performances of the engine. In a Stirling engine, the volume of the heater and regenerator are dependent and function of its power. These performances are function of the temperature ratio T_{paf}/T_{pah} .

It is very clear that the increase of dead space above the minimum required reduces the power parameter. Dead space in the heater would be reduced but the temperature drop between source and working fluid would be increased in inverse ratio. Further increase in efficiency can be obtained by maintaining the dead space as small as possible. Thus, the regenerator volume should not be small in order to ensure a good thermal change and it should not be large so that it does not represent a dead volume which penalizes the output of engine. Therefore, engine power increasing steadily as regenerator length is reduced. This is to be expected; power would be maximized by doing away with the regenerator altogether, provided enough heat could be supplied to make up for the resulting inefficiency.

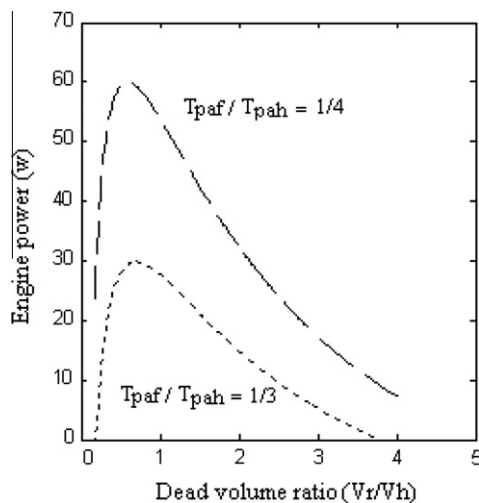


Fig. 16. Effect of the dead volume ratio on the engine power.

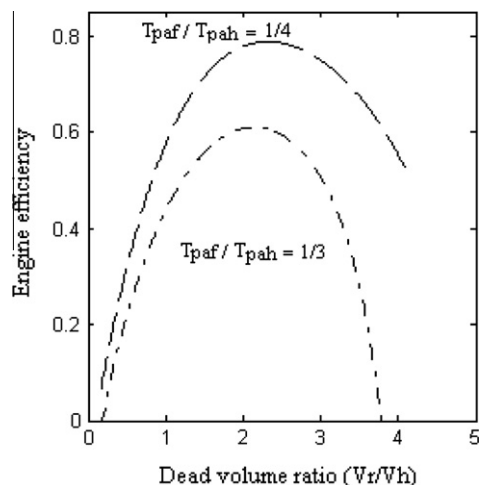


Fig. 17. Effect of the dead volume ratio on the engine efficiency.

Increase in dead volume within the Stirling engine system results in a loss of power but not necessarily a reduction in efficiency.

7. Conclusion

According to the performed literature survey, alpha Stirling engines using the Ross Yoke linkage drawn the minimum attention due to an old judgment that it has low experimental performance. However, it has many known advantages such as minimizing vibration and lateral forces acting on the pistons. Which lead to a more efficient and compact design with insignificant mechanical and thermal losses compared to other Stirling engines. In the present work, an attempt has been made to study this type of engine in order to improve its performance. Since this paper represents an application of the thermodynamic modeling in order to optimize the operating conditions of the Stirling engine, so that we obtain the best mode of power and performance. The parametric study of the engine performances highlighted the importance of the choice of some parameters (T_{pah} , M , τ , ϕ , ω) and their influence on the engine performance indices (Power, efficiency). We analyzed also the regenerator volume effect on the engine performance. The results obtained on the basis of the total model taking into account the thermal and mechanical losses show the fact that the regenerator parameters (thermal effectiveness, volume, losses) have a great impact on the performances of the Stirling engine. All these results are in agreement with the known experimental data for the Stirling engine [18,24,32,29,45,47].

The engine performance can be improved by increasing the precision of the engine parts and the heat source efficiency. The engine performance depends critically on the regenerator effectiveness and its ability to accommodate the high heat flow; it should be increased if a better working fluid e.g. helium or hydrogen is used instead of air.

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